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Journal Bearings with Hydrophobic Surface

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Abstract

In the article there is presented a modified Reynolds equation. The equation includes the effect of fluid slipping at partially hydrophobic surface. Bearing with the hydrophobic surface of the lining is characterized by a reduction of stiffness and damping for a maximum of one quarter of the original value in comparison with the hydrophilic lining. Significantly, however, decreases the value of dissipation function and thus the hydraulic losses in the bearing. In addition, increases the critical value of Taylor number.

Keywords: Hydrophobic element, hydrophilic element, sliding bearings, modified Reynolds equation, Taylor vortices.

1. Introduction

Journal bearings significantly affect the dynamics of the rotor. They are characterized by additional effects such liquid applied to the rotor of the machine. They are: additional weight, additional stiffness and additional damping tensor. The size of the elements of these tensors, namely stiffness and damping depend on the viscosity of the liquid.

The disadvantage of journal bearings is a high energy dissipation, which reduces the efficiency of the device. The simplest and economic are journal bearings with cylindrical lining. Their disadvantage is the instability of motion of the rotor at higher speeds and the necessity of adaptation to the critical value of Taylor number: $Ta < 41$, see term (1).

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$$Ta = \frac{\omega_1 \cdot R_1 \cdot s}{\nu \cdot \sqrt{\frac{s}{R_1}}} \leq 41 \,, \tag{1}$$

where: ω_1 is rotor angular speed, R_1 is shaft radius, s is radial gap, ν is kinematic viscosity.

The value of dissipation function can be significantly influent by the reduction of adhesive force between the bearing lining and the fluid. This reduction can be achieved by either partially or completely hydrophobic surface.

Under the new boundary conditions on the surface of the bearing lining, we deduce the modified shape of the Reynolds equation.

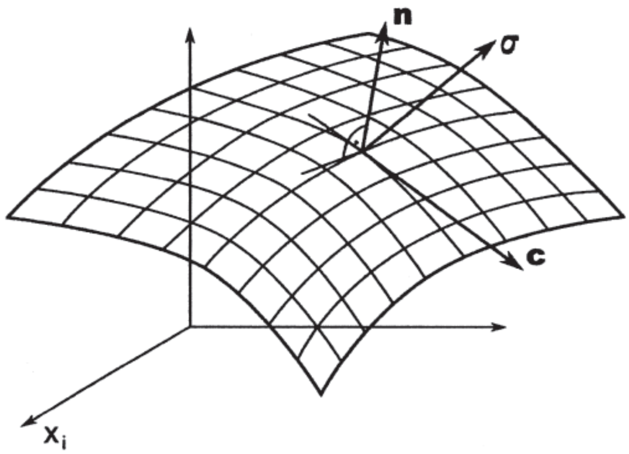

Fig. 1. 3 dimensional area

Assume that on the surface of bearing lining, it holds the following boundary condition [7]:

$$(\boldsymbol{\sigma}_A \times \mathbf{n}) \times \mathbf{n} = -k\mathbf{v} \,, \tag{2}$$

where: $\mathbf{v}$ - velocity vector, $\mathbf{n}$ - unit normal vector to the surface of the bearing lining, $\boldsymbol{\sigma}_A$ - viscous stress vector, k – adhesive coefficient.

For hydrophilic surfaces holds the condition $k \rightarrow \infty$, while for the hydrophobic surface it is $k = 0$.

Consider the 1D case of bearing shown in Fig. 2. Suppose partially wettable surface of the lining on the radius R_2. Then it is possible to write for the torque momentum the formula (2), from which it is obvious, the influence of the adhesive coefficient k. The value of k has significant influence on the creation of Taylor vortices and significantly shifts the critical value of Taylor number to higher values. On the following figures there is illustrated the influence of k on the number of Taylor vortices

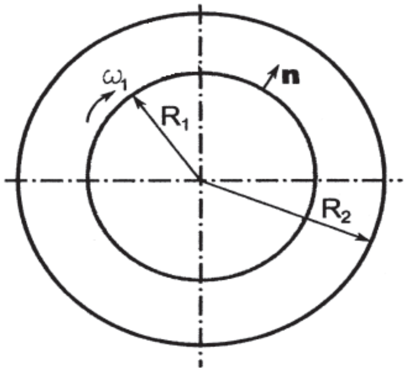

Fig. 2. Concentric cylinders

$$M_k = -\frac{4\pi U \eta R_1^2 R_2^2}{R_1\left(R_2^2 - R_1^2\right) + 2\frac{\eta}{k} R_2^2}. \quad (3)$$

where: η - dynamical viscosity, U - angular velocity, M_k - torque moment.

Table 1. Physical properties of liquids and geometric dimensions

Physical properties of liquid			
Liquid:	water		
Density	ρ =	998,2	kg.m^{-3}
Dynamical viscosity	η =	0,001003	Pa.s
Kinematical viscosity	ν =	1,00481E-06	m^2.s^{-1}
Geometric dimensions			
Outer diameter	D_1 =	50	mm
Inner diameter	D_2 =	30	mm
Gap size	s =	10	mm
Length of the ring (3D)	l =	120	mm

Table 2. Overview of the flow parameters for changing coefficient k (n = 52min^{-1})

k	***Number of vortices***	v_{ax}	v_{mag}	p_{max}	M_k	***WSS***
[-]	**[-]**	**[m.s^{-1}]**	**[m.s^{-1}]**	**[Pa]**	**[N.m]**	**[Pa]**
0,015	2	3,19E-04	3,40E-03	6,88E-04	1,33E-07	8,20E-04
0,2	8	4,34E-03	2,40E-02	2,83E-02	2,03E-06	1,25E-02
0,8	10	6,16E-03	4,64E-02	7,51E-02	5,10E-06	3,22E-02
no slip	10	8,68E-03	8,16E-02	6,59E-01	1,07E-05	7,10E-02

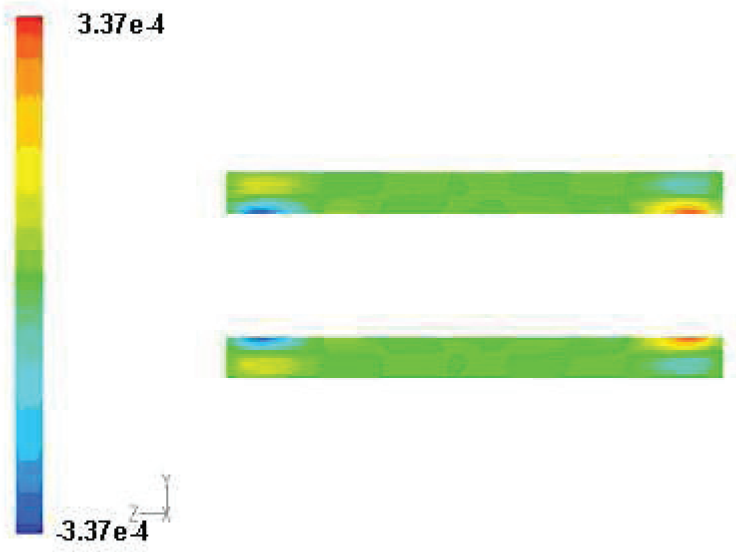

Fig. 3. Axial velocity ***k = 0,015***

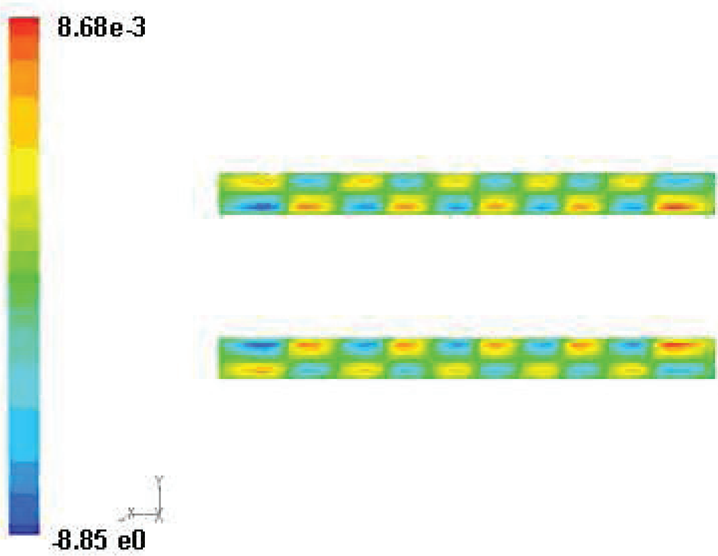

Fig. 4. Axial velocity ***no slip***

2. Modified Reynolds equation

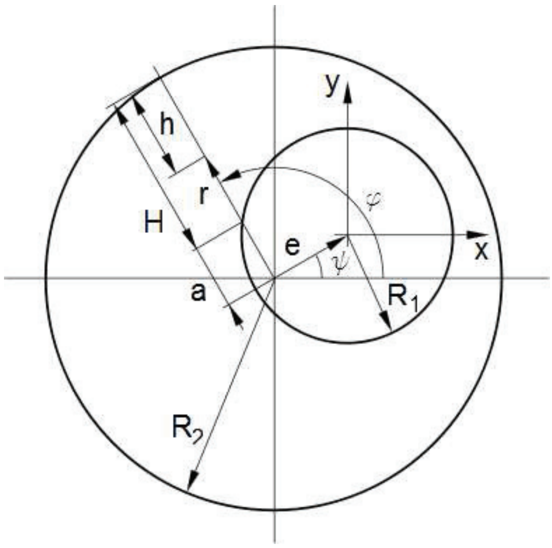

Fig. 5. Journal bearing

Assume that the shaft rotates with constant angular velocity ω_1. Furthermore, the rotor performs small oscillations around the equilibrium position, given by eccentricity e, that emerges from the static equilibrium of forces.

On the surface of the specified radius a, see Fig. 5, there hold the following boundary conditions for the speed $r = a$:

$$v_{\varphi a} = U - x^{\bullet} \sin\varphi + y^{\bullet} \cos\varphi \ , \tag{4}$$

$$v_{ra} = -U\left[\frac{x}{R_1}\sin\varphi - \frac{y}{R_1}\cos\varphi + \frac{e}{R_1}\sin(\varphi - \psi)\right] + x^{\bullet}\cos\varphi + y^{\bullet}\sin\varphi \ , \tag{5}$$

$$a = R_1 + y\sin\varphi + x\cos\varphi + e\cos(\varphi - \psi) \ , \tag{6}$$

$$H = R_2 - a \ , \tag{7}$$

where: x, y - components of displacement vector, $v_{\varphi a}, v_{ra}, x^{\bullet}, y^{\bullet}$ - components of the velocity vector in the circumferential, radial direction, e - eccentricity, dot is a symbol of the derivative by time.

On the partially wettable surface $r = R_2$ is due to (1):

$$\tau_{r\varphi} = -kv_{\varphi 0} \ , \tag{8}$$

where: $\tau_{r\varphi}$ - shear stress, $v_{\varphi 0}$ - circumferential velocity component.

Considering the simplified Navier-Stokes equation in the form:

$$\frac{\partial^2 v_{\varphi}}{\partial h^2} = \frac{1}{\eta R_2}\frac{\partial p}{\partial \varphi} \ . \tag{9}$$

And continuity equation:

$$\frac{\partial}{\partial \varphi}\int_0^H v_{\varphi} dh = av_{ra} - \frac{\partial a}{\partial \varphi} v_{\varphi a} \ . \tag{10}$$

If we denote:

$$\alpha = \left(1 + \frac{R_2}{a}\frac{k}{\eta}H\right)^{-1} \ , \tag{11}$$

where: η - dynamical viscosity.

We obtain the modified Reynolds equation in the form:

$$-\frac{\partial}{\partial\varphi}\left[\left(3\frac{R_2}{a}\alpha+1\right)H^3\frac{\partial p}{\partial\varphi}\right]=-6\eta R_2\frac{\partial}{\partial\varphi}\left[\left(\frac{R_2}{a}\alpha+1\right)Hv_{\varphi a}\right]+ \\ +12\eta R_2\left[av_{ra}-\frac{\partial a}{\partial\varphi}v_{\varphi a}\right]. \quad (12)$$

where: p - pressure.

3. Forces acting on the rotor in the journal bearing

$$F_y=-R_1L\int_0^{2\pi}\frac{\partial p}{\partial\varphi}\cos\varphi d\varphi;\qquad F_x=R_1L\int_0^{2\pi}\frac{\partial p}{\partial\varphi}\sin\varphi d\varphi. \quad (13)$$

When we express p, from the equation (12) we get:

$$\frac{\partial p}{\partial\varphi}=\frac{12_\eta R_2}{H^3\left(3\frac{R_2}{a}+1\right)}(N+f), \quad (14)$$

where:

$$N=(a+1)\frac{H}{2}c_{\varphi a}+\int\left(ac_{ra}-\frac{\partial a}{\partial\varphi}c_{\varphi a}\right)d\varphi\ , \quad (15)$$

$$f=-\int_0^{2\pi}\frac{N}{(3\alpha+1)}\frac{1}{H^3}d\varphi\Big/\int_0^{2\pi}\frac{1}{(3\alpha+1)}\frac{1}{H^3}d\varphi. \quad (16)$$

Using expressions (1), (4), (5), (12), (13), (14), (15) (16) can be determined matrixes of stiffness and damping.

In the case of $k\to\infty$: $a=0$, (12) goes in the classical Reynolds equation. For a hydrophobic surface, however $k=0$: $a=1$.

If $\frac{R_2}{a}\doteq 1$, the stiffness and damping of fluid layers are reduced on $\frac{1}{4}$ of the original values for the hydrophilic surface. But there occurs a significant decrease of dissipative function and increase the device power. Deficit of stiffness and damping can be compensate by reducing the bearing clearance while maintaining the reduced dissipative function compared to the original condition, or by increasing speed. For the qualitative assessment of the above mentioned conclusions will serve the following relations for the shear stress on the surface of the shaft.

Hydrophobic surface:

$$a=1;\ \ k=0:\ \ \tau_{r\varphi}=-\frac{H}{a}\frac{\partial p}{\partial\varphi}\ . \quad (17)$$

Hydrophilic surface:

$$a = 0; \quad k \to \infty: \quad \tau_{r\varphi} = \eta \frac{R_2}{a} \frac{c_{\varphi a}}{H} - \frac{1}{2} \frac{H}{R_2} \frac{\partial p}{\partial \varphi}. \tag{18}$$

We note that for the stationary case $e = 0,$ is

$$\frac{\partial p}{\partial \varphi} = 0. \tag{19}$$

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